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ScienceDirect
Energy Procedia 75 (2015) 1728 – 1733
Energy Procedia

The 7th International Conference on Applied Energy – ICAE2015

# Performance evaluation of displacement ventilation system combined with a novel evaporative cooled ceiling for a typical office in the city of Beirut

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**Abstract**

The study investigates by modeling the performance of displacement ventilation (DV) aided with a Novel Evaporative Cooled Ceiling, Maisotsenko cycle (M-cycle). The proposed system will increase the load removal of the DV system beyond the 40 W/m$^2$ limit with no additional energy consumption. Predictive mathematical models of the conditioned space and the evaporative cooled ceiling will be developed to study the energy performance of the suggested combined system for typical offices of Beirut. The study identifies different values of supply air relative humidity at a certain supply flow rate and temperature while meeting space load, indoor air quality, and thermal comfort. The proposed system resulted in cooling load removal capacity enhancements of 18.65%, 44.3% and 72.25% at supply air relative humidity of 90%, 50% and 10°C respectively. The results indicated a better performance at lower supply air relative humidity.


Peer-review under responsibility of Applied Energy Innovation Institute

*Keywords:*Displacement ventilation; Maisotsenko cycle; indoor air quality; mathematical modeling

## 1. Introduction

Human beings currently spend 90% of their time in buildings and 60% of world-wide energy produced is spent in residential buildings [1], hence the need to reduce energy consumption. One of the promising air conditioning systems considered to provide both thermal comfort and air quality with low energy consumption is displacement ventilation (DV) [2] and [3], but the major issue about this system is the limit of low cooling loads of about 40 W/m$^2$. For this reason researchers have suggested using the chilled ceiling displacement ventilation (CC/DV) system to increase the load removal of DV systems up to 100

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Peer-review under responsibility of Applied Energy Innovation Institute
doi:10.1016/j.egypro.2015.07.440

W/m$^2$. However, this enhancement in load removal came at the expense of additional energy consumption. So passive ways need to be used to enhance the performance of the DV system and increase its load removal using evaporative cooling of the ceiling. In this study, the chilled ceiling model incorporates a Maisotsenko cycle (M-cycle) by passing the DV upper air through a novel evaporative channel to lower the temperature of the ceiling and assist in removing additional space sensible load.

**Nomenclature**

| | |
|---|---|
| a | channel height (m) |
| $q'_R$ | heat load (W/m$^2$) |
| T | temperature (◦C) |
| X | humidity ratio (kg/kg) |
| X* | humidity ratio of saturated air (kg/kg) |
| z | position (m) |
| α | convective heat transfer coefficient (W/m$^2$.K) |
| αm | mass transfer coefficient (m/s) |
| $K_{e1}$, $K_{e2}$ | effective thermal conductivity of the heat transfer plate and ceiling (W/m.K) |
| $a_{e1}$, $a_{e2}$ | effective thickness of the heat transfer plate and ceiling (m) |
| *Subscripts* | |
| d | dry channel |
| p | heat transfer plate |
| r | room |
| s1, s2 | water absorbing sheet |
| e1, e2 | effective value corresponding to the transfer plate and the ceiling |
| w | wet channel |

## 2. Performance analysis of displacement ventilation system combined with a novel evaporative cooled ceiling

### *2.1. Description of the proposed system*

Cool supply air enters at floor level and picks the load from the space as shown in Fig 1(a). After that the air is directed toward the ceiling where it passes in a dry channel and gets cooled down without gaining any humidity. Then it passes in a counter flow manner through the wet channel where it gains humidity through the evaporation of water and decreases the temperature of the ceiling as shown in Fig 1(b). The office, located in Beirut, has an 8.2 m$^2$ floor area and a height of 2.8 m. It consists of one exposed wall on the south facade. The remaining walls along with the floor are partitioned with conditioned spaces. The internal heat in the office has a sensible load of 40 W/m$^2$.

### *2.2. Mathematical modeling*

To predict the performance of the combined system, a mathematical model for the DV system and the evaporative cooled ceiling are needed.

#### *2.2.1. DV Model*

The widely used DV plume-multi-layer thermal space model of Ayoub et al. [4] is adopted. The DV model predicts indoor air temperature for given internal loads and outdoor ambient conditions, under certain supply air conditions.

#### *2.2.2. Evaporative Cooled Ceiling Model*

To predict the ceiling temperature different energy and mass balance equations were developed using the study of Miyazaki et al. [5].

- Energy balance for the air in the dry and wet channels

$$\rho_d C_{pd} u_d a_d \frac{dT_d}{dz} = \alpha_d \left(T_p - T_d\right) \tag{1}$$

$$\rho_w C_{pw} u_w a_w \frac{dT_w}{dz} = \alpha_{w1} \left(T_{s1} - T_w\right) + \alpha_{w2} \left(T_{s2} - T_w\right) \tag{2}$$

The right term of equation 1 and 2 represent the net convective heat flow in the dry and wet channels respectively. The left hand side of equation 1 represents the heat exchange in the dry channel. The left hand side of equation 2 represents the heat exchange with the two sides of the wet channel.

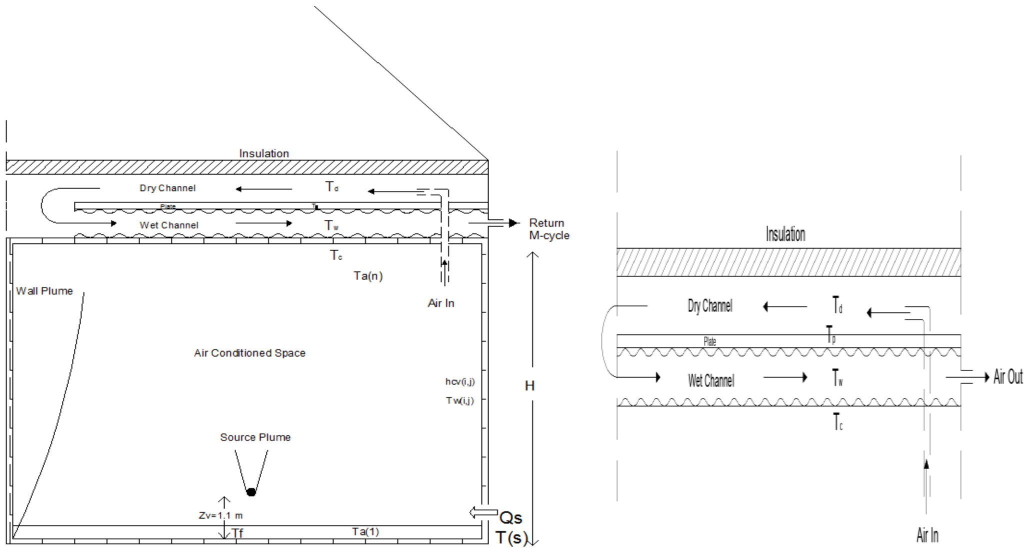


Fig. 1. (a) Schematic of the combined air conditioning system; (b) Schematic of the ceiling M-cycle

- Energy balance of the two water sheets

$$\frac{K_{e1}}{a_{e1}}\left(T_p-T_{s1}\right)+\alpha_{w1}\left(T_w-T_{s1}\right)+\rho_d\alpha_{m1}h_{fg}\left(X_w-X_w^*\right)=0 \tag{3}$$

$$\frac{K_{e2}}{a_{e2}}\left(T_c-T_{s2}\right)+\alpha_{w2}\left(T_w-T_{s2}\right)+\rho_d\alpha_{m2}h_{fg}\left(X_w-X_w^*\right)=0 \tag{4}$$

In equation 3, the upper water sheet exchanges heat with the plate and exchanges sensible and latent heat transfer with the flowing air. Similarly, in equation 4 the lower water sheet exchanges conduction heat transfer with the ceiling plate and sensible and latent heat with the flowing air.

- Mass balance of the air in the wet channel

$$\rho_d u_w a_w \frac{dX_w}{dz}=\rho_d\left(\alpha_{m1}+\alpha_{m2}\right)\left(X_w^*-X_w\right) \tag{5}$$

The right hand side of equation 5 represents the net convective flow of moisture in the wet channel and the left hand side represents the moisture exchanges with the two water sheets of the wet channel.

- Energy balance of the ceiling and the plate separating the wet and dry channels

$$\frac{K_{e2}}{a_{e2}}\left(T_{s2}-T_c\right)+\alpha_r\left(T_r-T_c\right)+\dot{q}_R=0 \tag{6}$$

$$\alpha_d\left(T_d-T_p\right)+\frac{K_{e1}}{a_{e1}}\left(T_{s1}-T_d\right)=0 \tag{7}$$

The ceiling exchanges heat with the water sheet of the wet channel in equation 6 as well as convective and radiative heat transfer with the space. The first left term in equation 7 represents the heat transfer with the dry air and the second term represents the conductive heat flow with the water sheet.

### *2.3. Integrated model sequence of operation*

In order to solve the system of equations for the vertical temperature distribution in the space and the system capacity for load removal, the required inputs are the air supply flow rate and temperature, the outdoor weather conditions such as solar radiation and ambient conditions and the internal heat loads, where the software used to solve the equations is Matlab. Internal surface temperatures found by the DV model will be used by the evaporative cooled ceiling model to calculate the load subjected to the ceiling and update the ceiling temperature. Then, the updated temperatures are used to solve for the energy balance of each air layer. This shows that the solution requires solving the different models iteratively, in order to get at each time step the different variables and update them until convergence is reached. The coupled mass and energy equations are discretized into algebraic equations using the finite volume method. Once convergence is reached the temperature distribution in the space is determined as well as the energy performance of the system. The convergence condition for the residuals of the different equations is set to $10^{-6}$.

## 3. Results and discussion

The models were simulated at supply air flow rate of 0.14 $m^3/s$ and supply air temperature of 22°C with 10%, 50% and 90% relative humidity. Simulations were done for the model when using the DV system alone and other simulations were done for the model when using the integrated system of the evaporative cooled ceiling and DV system. Then the room temperature was recorded for the different simulations as well as the total cooling load removed by the system. Comparison was done between the two systems to calculate the improvement in total load removal.

Fig 3(a) shows the variation of the total load removed in $W/m^2$ by the DV system and the integrated evaporative cooled ceiling and DV system at different supply air relative humidity. Results showed the load removed by the DV system was 40 $W/m^2$ and it was constant for the different relative humidity values, however; the load removed by the integrated system was maximum and equal to 68.9 $W/m^2$ at the lowest relative humidity of 10% and it decreased to reach 47.46 $W/m^2$ at the highest relative humidity of 90%. The improvement in sensible load removal when using the proposed integrated system was 18.65%, 44.3% and 72.25% at relative humidity of 10%, 50% and 90% respectively. Fig 3(b) shows the variation of the room air temperature when using the DV system alone and then when using the integrated system at different supply air relative humidity. The room temperature when using the DV system alone was 24°C and it was not affected by the change in the supply air relative humidity. When using the integrated system of the evaporative cooled ceiling and the DV system, the room temperature increased from 23.4°C to 23.8°C at the relative humidity of 10% and 90% respectively. This shows that the integrated system has better sensible load removal at lower supply air relative humidity.

## 4. Conclusion

Simulation results showed that the displacement ventilation system combined with a novel evaporative cooled ceiling can improve the displacement ventilation cooling capacity with no additional energy consumption. Mathematical modelling of the proposed system was performed. Results showed enhancements of the proposed system between 18.65% and 72.25% at supply air relative humidity between 10% and 90% respectively. Better cooling performance was attained at lower supply air relative humidity, which justifies the use of desiccant dehumidification systems to lower the humidity of supply air before it enters and cools the space.

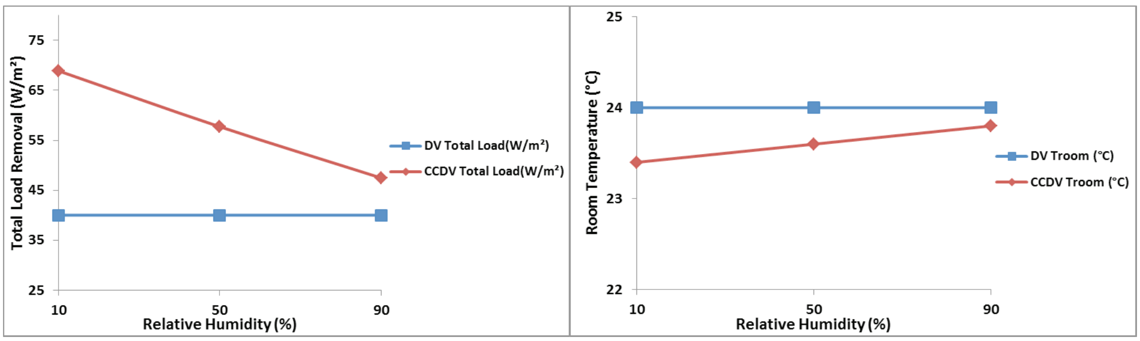


Fig. 3. (a) Total load Removal at 10, 50 &90% relative humidity; (b) Room temperature at 10, 50 &90% relative humidity

## Acknowledgements

Authors would like to express their gratitude to the Qatar Chair Fund for the support to conduct this study.

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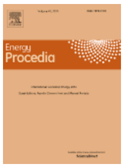


## Biography

Eng. Mariam Itani is a research assistant at American University of Beirut working on displacement ventilation system combined with a novel evaporative cooled ceiling. She graduated a mechanical engineer from Beirut Arab University in 2013 and is expected to get her Masters degree from American University of Beirut in 2015.